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PERFORMANCE OF THE SNAP-8 LUBRICATION AND COOLANT SUBSYSTEM

by H. B. Block, R. Kruchowy, and J. D. Gallagher

Lewis Research Center
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ABSTRACT

A SNAP-8 system containing all major prototype components except the nuclear reactor and the radiator was assembled and tested for 1445 hours at Lewis Research Center. The overall system consisted of five subsystems. One subsystem lubricated and/or cooled the other four SNAP-8 subsystems.

Lubrication and coolant subsystem pump parameters, and component endurance data were examined. Data indicated no degradation in pump performance during the test. Fluid was supplied to the SNAP-8 components at the required flow rate and temperature. None of these components experienced a failure or significant degradation caused by lack of lubrication or cooling. Proper cooling of the mercury pump and the turbine alternator space seals resulted in a very low oil and mercury leakage to the space vacuum simulator.

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SUMMARY

A SNAP-8 system containing all major prototype components except the nuclear reactor and the radiator was assembled and tested for 1445 hours at Lewis Research Center. The overall system consisted of five subsystems. One subsystem lubricated and/or cooled the other four SNAP-8 subsystems.

Lubrication and coolant subsystem pump parameters, and component endurance data were examined. Data indicated no degradation in pump performance during the test. Fluid was supplied to the SNAP-8 components at the required flow rate and temperature. None of these components experienced a failure or significant degradation caused by lack of lubrication or cooling. Proper cooling of the mercury pump and the turbine alternator space seals resulted in a very low oil and mercury leakage to the space vacuum simulator.

INTRODUCTION

Future space flights will require greater amounts of electrical power than is presently used. One potential system being developed for these requirements is the SNAP-8, Rankine cycle electrical generating system which can produce 35 kilowatts of usable electrical power. A SNAP-8 system incorporating all major flight components was assembled at the Lewis Research Center to obtain steady state and endurance data.

The overall system consisted of five main subsystems: the primary or heat source subsystem which utilized a eutectic mixture of sodium-potassium (NaK) as the heat transfer medium; the power subsystem which used mercury as the working fluid; the heat-rejection subsystem which used NaK to remove waste energy from the power subsystem; the lubrication and coolant subsystem which utilized an organic fluid, 4P3E polyphenyl ether, to lubricate and/or cool four SNAP-8 flight components; and the space seal vacuum subsystem which removed oil and mercury leakage from the dynamic fluid seals of the mercury pump and the turbine alternator.

The performance of any one subsystem directly affects all other subsystems. To gain confidence and verify the capability to meet the SNAP-8 system objective of demonstrating 10 000 hours of continuous operation, many hours of system testing are required. Therefore, an endurance test lasting 1445 hours was conducted at the Lewis Research Center. The purpose of this publication is to report the performance of the lubrication and coolant subsystem during this endurance test. Pertinent pump parameters and subsystem endurance data are presented.

Lubrication and coolant pump performance curves are presented in reference 1. References 2 to 6 contain design information, and performance analysis of the remaining SNAP-8 subsystems and major components during this endurance test.

DESCRIPTION

The lubrication and coolant system (or the oil system) schematic is shown in figure 1. A photograph of the overall system is shown in figure 2. The lubrication and coolant pump circulated mix 4P3E polyphenyl ether (oil) through two parallel heaters, an oil to water heat exchanger, and two parallel dual filters to a manifold. Oil was distributed from the manifold to four SNAP-8 components. Oil was returned from the components to the pump through an expansion reservoir, a mercury trap, and a screen filter.

The two parallel heaters, along with the line heaters, were used during system startup to preheat the oil. After the SNAP-8 components became operational, the oil to water heat exchanger was used to remove waste energy from the lubrication and coolant system. Oil discharge temperature from the oil to water heat exchanger was maintained within 10° F (5.6 K) by a temperature sensitive electropneumatic controller. The parallel filters were capable of removing particles down to 10 microns (1×10^{-5} m) nominal from the system.

The expansion reservoir was installed at the high point of the system to take up any oil expansion, and for degassing purposes. During normal system operation, the expansion reservoir was maintained at a vacuum of about 0.050 torr (6.67 N/m^2). As oil passed through the reservoir, trapped gas was removed. This type of oil was susceptible to gas entrapment and had to be handled in the above manner.

The reasons for selecting mix 4P3E polyphenyl ether for the SNAP-8 lubricant and coolant subsystem and problems encountered with the fluid are discussed in reference 7. Generally, this oil is a radiation resistant lubricant with a low vapor pressure that can withstand hot-spot temperatures up to 700° F (644 K).

The majority of the oil system plumbing was comprised of stainless steel 1-inch (2.54×10^{-2}-m) diameter tubing. Glass fiber molded insulation was used on all straight

tubing runs, while blanket type insulation was used on the remainder of the system. Insulation thickness averaged about 2 inches (5.08×10^{-2} m).

Lubrication and Coolant Pump

The lubrication and coolant pump (oil pump) consisted of a single shaft with a straddle-mounted motor rotor and an overhung single-stage impeller (figs. 3(a) and (b)). This unit was self cooled and lubricated by system oil. Oil bleeds from the pump discharge, flows through the motor and bearing area, and returns to the eye of the impeller through the hollow shaft. Both the carbon thrust and journal bearings, and the ML (polyimide) insulation of the motor are compatible with the system oil. Figure 3(c) shows the oil pump installation. The pump was installed in this manner to obtain maximum suction head (NPSH).

Components Serviced by the Oil System

Component areas serviced by the oil system were as follows:

(1) Mercury pump
- (a) Motor heat exchanger
- (b) Bearings
- (c) Space seal heat exchanger

(2) Primary loop NaK pump heat exchanger

(3) Heat rejection loop NaK pump heat exchanger

(4) Turbine alternator
- (a) Turbine space seal heat exchanger
- (b) Bearings
- (c) Alternator heat exchanger

Mercury pump. - Reference 3 contains a detailed description of the mercury pump. Waste energy was removed from the mercury pump motor heat exchanger by the oil passing through the annuli in the motor housing. The oil was then filtered and directed to the mercury pump ball bearings. Inlet and discharge lines to this component had a static head of about 10 feet (3.05 m) of oil. Two valves in series were installed on these lines to prevent accidental oil leakage through the pump into the mercury system when it was not running. Internal leakages were prevented when the pump was not rotating by two bellow-actuated carbon face seals, sealing against opposite surfaces of a rotating ring. Prevention of mercury system contamination was very important because a small amount of oil mixed with the mercury can cause deconditioning of the boiler and a

substantial drop in system performance (ref. 8).

When the mercury pump was rotating, a slinger located near the ball bearings pumped oil out of the bearings cavity into the discharge line (fig. 4). Molecular boiloff from the slinger liquid interface was contained by a molecular pump which consisted of helical grooves located around the pump's rotating shaft. A vacuum was used to simulate the space environment in the area between the oil and mercury seals.

The mercury seals were similar to the oil seals. In this case, mercury leakage down the shaft to the oil system was sealed by a viscopump, a molecular pump, and finally the common vacuum area. This combination of mercury seals, common vacuum, and oil seals is called the "space seal." Rated oil flow was maintained in the mercury pump space seal heat exchanger during this test. This was done to maintain a low local fluid temperature and thereby minimize leakage and cavitation damage to the mercury viscopump.

Primary and heat rejection loop pumps. - Primary and heat rejection loop pumps (ref. 4) contained trapped NaK volumes which were used to cool the motors and lubricate the bearings. A small internal pump circulated the trapped NaK through the motor housing, the bearings, and into a heat exchanger. Waste energy and NaK oxides were removed in the heat exchangers. Again, oil was supplied in large quantities in order to maintain a low average temperature in each unit.

Turbine alternator. - Turbine space seal design was similar to that of the mercury pump (fig. 5). The major difference in the turbine was that mercury vapor had to be condensed first, then contained in the viscopump area. A high oil flow was required to remove the heat of condensation, and again to maintain a low regional temperature. Oil flowed in a series path through the turbine space seal heat exchanger into the alternator heat exchanger. Alternator losses were removed in this manner (ref. 5).

Turbine alternator ball bearings were lubricated by a small oil flow rate. To prevent oil leakage through the bearings into the mercury system when this unit was not rotating, double valves in series were incorporated on the inlet and discharge lines.

Two other flow lines were built into this system. The first, a jet pump line, was intended to maintain flow through the mercury pump motor heat exchanger and bearings if necessary. At a later date, it was determined that flow was sufficient and that the jet pump action was not required. The second, a bypass line, was used to obtain pump head curves.

INSTRUMENTATION

Instrumentation used in documenting system performance consisted of thermocouples, pressure transducers, and turbine flow meters as shown in figure 1. Additional

instrumentation included a speed sensor and a power measuring circuit.

Several internal thermocouples (chromel-alumel) were supplied with the lubrication and coolant pump to measure winding hot-spot temperature. Iron-constantan thermocouples (Instrument Society of America type J), referenced to 32° F (273 K) junctions were used to measure system temperatures. These thermocouples were spot welded to the outside tube surfaces of the plumbing. Oil temperature values on the inlet and discharge of each component were used in the rejected energy (heat loss) calculations. The mercury pump space seal heat exchanger, primary pump and heat rejection pump heat exchangers rejected energies are to be considered as representative values because the temperature difference from inlet to discharge was a maximum of 4° F (2.2 K).

Pressure measurements were made using commercial, stainless steel, variable reluctance type transducers. This type of transducer operated linearly up to 250° F (394 K). To check the output of the transducers, small Bourdon gages were installed on the oil pump inlet, discharge, and on the dump tank.

Turbine flow meters were used to measure oil flow rates throughout the system. Standard commercial models were selected and welded in the various locations.

The oil pump was supplied with a built-in variable reluctance magnetic speed sensor. Oil pump voltage, current, and power were measured with average sensing transducers. These transducers were designed for a 400 Hz pure wave form at a minimum power factor of 0.7. Accuracy of the power measurements was only ± 10 percent because the alternator did not produce a 400 Hz pure wave form, and the oil pump ran slightly lower than 0.7 power factor.

A thermocouple-type vacuum probe was used to measure expansion reservoir pressure during system evacuation and normal operation.

All data presented in this report were recorded on magnetic tape using an automatic high-speed digital recording system (CADDE, Central Automatic Digital Data Encorder, ref. 9). Various pertinent system parameters were monitored on a central control panel shown in figure 6.

PROCEDURE

Calibration

All thermocouples received a continuity and heat check. The checks were made to ensure proper thermocouple lead connection and response to temperature.

Pressure transducers were calibrated before installation and again prior to the test. Preinstallation calibrations were made to check signal output, hysteresis, and repeatability. Calibration prior to the test was performed in a similar manner then correction

curves were plotted. Reference pressures were measured with a precision Bourdon gage (0.1 percent of full scale accuracy).

Turbine flow meter signal conditioning equipment were calibrated using a $\pm 1/2$ percent variable frequency source. Pump speed monitoring equipment, and the electrical monitoring equipment were calibrated in a similar manner.

Cleaning

Prior to the final oil system buildup, all components located on the stand (fig. 2) were cleaned with a benzene flush. A backup pump was used to circulate benzene until remaining particle size was no larger than a nominal 10 microns (1×10^{-5} m).

Lines connecting the stand to the SNAP-8 components were precleaned prior to installation. A total system cleaning was then performed using a backup pump to circulate freon. Pumping continued until remaining dirt in the system was again 10 microns (1×10^{-5} m) nominal in partical size. The freon was then drained, and the system was evacuated through the expansion reservoir down to 0.040 torr (5.33 N/m^2). At no time was cleaning fluid allowed into the mercury pump or turbine alternator ball bearing areas.

Operation

The dump tank was charged with mix 4P3E after system cleaning. Dump tank heaters were turned on, and the oil was gradually brought up to 200° F (366.6 K). A vacuum of below 1 torr (133.3 N/m^2) was applied to the dump tank. After two weeks, the oil was degassed and the vacuum level was maintained at 0.050 torr (6.67 N/m^2).

Oil was then forced into the system until the expansion reservoir was half full. Pressure in the expansion reservoir at that time was about 1 torr (133.3 N/m^2). System heaters were turned on, and the oil pump was started. Oil temperature was gradually increased to about 220° F (377.6 K) at the pump discharge. The system was degassed for 24 hours prior to the overall SNAP-8 system startup. Expansion reservoir pressure was maintained between 0.50 torr (6.67 N/m^2) and 0.10 torr (13.33 N/m^2) during the entire test.

Oil flows to most components were preset just before the SNAP-8 system startup occurred. However, no flow was directed through the mercury pump bearings until that pump was started, and had attained rated speed of 5800 rpm (ref. 3). Also, the turbine alternator bearings received no oil flow until the assembly reached 9600 rpm (ref. 5).

Oil system filters were changed once during the test. The elements were reassem-

bled so air would not enter the system. To accomplish this, each filter bank was evacuated down to 0.010 torr (1.33 N/m^2) through a valve located on the filter housing. Then the filter bank was reopened to the system.

RESULTS AND DISCUSSION

Oil Pump Performance

The oil pump accumulated a total of 1584.4 hours of operation (1445 hours in conjunction with the overall SNAP-8 system test), and experienced 47 starts and stops during that time. Pump head, power, and efficiency data, presented in figure 7, were obtained prior to and after the system test. Also shown in figure 7 are data (reference data) obtained from the SNAP-8 prime contractor.

Pretest, post-test, and reference pump head rise data (fig. 7(a)) were in agreement indicating that the pump head curve did not change during the test.

Input power and overall pump efficiency against volumetric flow rate are shown in figure 7(b). Pretest and post-test data are shown as an average curve because power monitoring equipment was of limited accuracy as discussed before. Reference power should agree with prerun data because the pump was not run from the time of original checkout until this test. However, the two curves differed by about 10 percent even though the contractor used similar power measuring equipment. Instrument inaccuracy was the probable cause for differences in the power curves.

The oil pump overall efficiency is defined as:

$$\text{Pump Overall Efficiency} = \frac{\text{Pump Hydraulic Power}}{\text{Electrical Input Power}} \times 100 \text{ Percent}$$

where

$$\text{Pump Hydraulic Power} = \frac{\text{Volumetric Flow Rate} \times \text{Total Dynamic Head Rise}}{\text{Constant}}$$

Prerun and post-run data indicated no change in efficiency during the test. At the normal operational point of about 13 gpm (0.049 m^3/min), overall pump efficiency was about 26 percent. This low efficiency was due to the oil bleed from the pump discharge used to self cool the windings, and lubricate the pump bearings. Reference efficiency generally ran higher because it was based on a lower input power.

Pump electrical endurance parameters are shown in figure 8(a). During the first

8 days of the test, pump power was supplied by a facility source. This power supply generated a higher phase voltage (about 5 V) and a higher frequency than the alternator which normally supplied the pump. Because phase voltage and supply frequency were higher during the first days of operation, the pump ran from 100 to 200 rpm faster than normal. As a result, total oil flow rate was greater than normal during this time period. From the ninth day, until the conclusion of the test, pump electrical parameters changed very little. Pump efficiency, speed, and supply frequency varied less than ±1/2 percent of the average value for each parameter. Input power, average phase amperage, and average phase voltage varied less than ±1.7 percent.

Figure 8(b) presents the pump mechanical endurance parameters. Variations were observed in discharge and suction pressures. Generally, changes in these pressures reflect changes in total oil flow rate. A good example of this occurred when the line filters were changed. Total oil flow increased almost 200 pounds per hour (90.8 kg/hr), while the discharge and suction pressures dropped accordingly. On the 44th day of the test, total flow began falling off. At the same time discharge and suction pressures increased. This was probably due to an accumulation of dirt particles in the line filters and a corresponding increase in pressure drop. Total oil flow rate (from the ninth day) averaged about 7400 pounds per hour (3355 kg/hr). Variations amounted to ±1.5 percent of the average value during the remainder of the test.

Pump suction pressure was maintained at 2.1 psia (14.48×10^3 N/m^2) or greater during the test. Minimum NPSH for this pump was 1.6 psia (11.03×10^3 N/m^2). The NPSH was well above this minimum value, and the unchanged pump head curve indicates that the impeller probably did not suffer any cavitation damage.

No upward trends were evident in the pump housing temperature, oil temperature rise across the pump, and the oil discharge temperature.

These relatively stable pump temperatures, coupled with the previously mentioned constant electrical and pressure parameters, indicate no internal problems were developing in the pump.

Oil System Performance

The data obtained for each of these components were evaluated to determine the effectiveness of the oil system in lubricating and cooling these components.

Mercury pump. - Lubrication and coolant system oil lubricated the mercury pump bearings, and cooled the motor heat exchanger and space seal heat exchanger. Figure 9(a) presents the two mercury pump component oil flow rates against time. Both flows generally responded with variations in total oil flow. Minor variations in flow were caused by small position changes in the control valves.

Mercury pump bearings and motor heat exchanger oil flow rate was maintained between 155 and 175 pounds per hour (70.4 and 79.5 kg/hr) throughout the test. Flow was maintained at this relatively low level because of the low component rejected energy levels (a total of about 1 kW), and the small flow required to lubricate the pump ball bearings (yet not flood the bearings). During several pump starts it was determined that oil flow to the bearings should be initiated after the pump had reached rated speed. When oil flow was initiated before rated speed was attained, the mercury pump speed decreased causing a current increase, and the circuit breakers disengaged.

Space seal heat exchanger flow rate varied from 2260 to 2450 pounds per hour (1025 to 1112 kg/hr) after oil pump supply power was transferred to the alternator. This flow was required to maintain a low temperature in the space seal area. The function of the space seal was to prevent intermixing of oil with mercury. Small amounts of oil and mercury were found in the space seal common vacuum area after the test. However, several chemical analysis conducted during and after the test indicated no contamination in either the oil or mercury system.

Mercury pump supply oil temperatures are shown in figure 9(b). Motor heat exchanger and bearing oil was supplied at about 180° F (355 K) throughout the test. Temperature instrumentation built into the pump indicated that at no time did the winding hotspot temperature exceed 335° F (441 K). Motor windings maximum limit was 400° F (478 K). Pump space seal heat exchanger oil was supplied at 195° F (364 K). This temperature was well within a recommended normal operational limit of 300° F (422 K).

Rejected (or waste) energy data for this pump are presented in figure 9(c). Liquid mercury flow rate through the pump is also shown. The motor heat exchanger and bearings, and the space seal heat exchanger rejected energy curves varied little during the test. The average energy level was 0.5 kilowatt for each parameter. No trends in rejected energy were apparent in either case even when the mercury flow rate was changed.

Examination of the above data, the mercury pump head curve, and the input power requirements (ref. 3) indicated no deterioration in pump performance. The lubrication and coolant system apparently serviced the mercury pump satisfactorily.

Primary and heat rejection loop NaK pumps. - The oil system parameters pertinent to the NaK pumps are presented in figure 10. Lubrication and coolant oil flow to the NaK pumps against time is shown in figure 10(a). Oil flowed through the heat exchangers, which were located in series, on the primary loop and the heat rejection loop pumps. The heat exchangers were used in the purification (oxide removal) and cooling of a small volume of trapped NaK which was used to lubricate and cool the pump's bearings. Maintaining a uniform and relatively cool oil temperature in the heat exchangers assures maximum oxide removal. For this reason, the primary and heat rejection loop pumps' oil flow was controlled at about 2300 pounds per hour (1043 kg/hr). Variations in component flow rate were due to changes in total oil flow rate.

Oil inlet temperatures against time for the NaK pump heat exchangers are shown in figure 10(b). Oil was supplied to the primary NaK pump heat exchanger at about 200° F (366 K), and to the heat rejection NaK pump heat exchanger at 205° F (369 K). These values are well below the maximum limit of 300° F (422 K).

Figure 10(c) shows the rejected energy curves for the NaK pump heat exchangers. The primary loop and the heat rejection loop pump heat exchangers averaged 1.2 kilowatt rejected energy each during the test. The rejected energy for each NaK pump was stable throughout the test. An examination of NaK pump data (ref. 4) over the 1445 hour endurance test indicated no change in their performances. Because no change in pump performance was detected, the assumption was made that the oil system serviced the NaK pumps as designed.

Turbine alternator. - One common line supplied oil to two parallel paths on the turbine alternator. The first path lubricated the assembly bearings, and the second cooled the heat exchanger. The heat exchanger line cooled two sections connected in series: the turbine space seal heat exchanger, and the alternator heat exchanger.

Figure 11(a) shows the oil flow rates in the two turbine alternator parallel paths. As in the case of the mercury pump, the bearings flow was maintained at an average value of 830 pounds per hour (376 kg/hr). Turbine alternator and mercury pump bearing designs are similar. Flow to the turbine alternator bearings was initiated after a speed of 9600 rpm was attained. The flow to the heat exchangers averaged 1725 pounds per hour (783 kg/hr). This higher flow was required to maintain an average low temperature in the space seal area, and the alternator windings.

Major changes in turbine alternator oil flows were again caused by changes in total oil flow rate. The bearings flow, however, had secondary perturbations which were caused by changes in position of the discharge valve.

Turbine alternator oil supply temperature (fig. 11(b)) averaged about 200° F (366 K) during the test. This value was well within recommended normal operational limits. Internal temperature instrumentation indicated that none of the four bearing races exceeded recommended maximum value of 300° F (422 K). The turbine-end bearing temperature, however, increased near the end of the test to nearly 300° F (422 K).

Figure 11(c) presents the turbine alternator rejected energy curves. Bearings rejected energy varied ± 0.5 kilowatt about an average of 5 kilowatts until the 45^{th} day of the test. Starting on this day, the rejected energy level increased slowly. Disassembly and inspection of this unit after the test showed evidence that the turbine-end bearing had been operated for an extended period with a large load unbalance (ref. 5). Wear caused by the unbalanced bearing loading probably caused the gradual increase in rejected energy starting on the 46^{th} day.

Reference 5 gives a detailed discussion of the alternator and space seal heat exchangers rejected energies. Alternator heat exchanger rejected energy was about 0.7 kil-

owatt at the beginning of the test. When all SNAP-8 system pumps were transferred from facility power to alternator power, alternator heat exchanger rejected energy increased to a new level of about 2.0 kilowatts. This increase occurred because the system pumps had low power factors and high current requirements as compared to the resistor load bank, which absorbed all alternator power prior to the pump power transfer. Higher current flow in the alternator windings, therefore, caused the increase in rejected energy. Alternator winding hot-spot temperature did not exceed 360° F (455 K). Maximum allowable temperature for this component was 500° F (533 K). An electrical check made after the test confirmed that all alternator windings were normal. The conclusion was that the oil system cooled the alternator as designed.

Mercury flow rate was at the highest level at the beginning of the test (fig. 11(c)). Also, turbine alternator mercury discharge pressure was at a higher level than normal (ref. 5). This high discharge pressure resulted in a high mercury temperature near the turbine space seal. As expected then, turbine space seal rejected energy was greater at the beginning of the run. As the test progressed, mercury flow was gradually decreased. Turbine discharge pressure was also decreased, resulting in a decrease in turbine rejected energy level. Traces of oil and mercury were found in the space seal common vacuum area after the test. Chemical analysis, however, indicated that no contamination was present in the mercury or oil systems. Therefore, the conclusion was that the oil system serviced the turbine space seal heat exchanger properly.

A summary of typical average rejected energy levels for the components served by the oil system is presented in table I.

CONCLUSIONS

A 1445-hour endurance test was conducted on the overall SNAP-8 system using all major flight components. Performance of the lubrication and coolant (oil) subsystem, including the pump was studied.

Conclusions on the performance of the oil subsystem were as follows:

1. No changes in oil pump mechanical or electrical parameters were detected that indicated any degradation in pump performance.

2. Oil was supplied to the SNAP-8 components at the design flow rate and temperature.

3. No failures were experienced in any SNAP-8 component which were attributable to the oil subsystem. However, one turbine alternator bearing was damaged during the test. Inspection indicated that the bearing damage was caused by a large load unbalance applied for an extended period of time, not by insufficient lubrication.

4. Leakages through the mercury pump and the turbine alternator space seals were maintained at a low level by proper oil cooling during the test.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, November 1, 1968,
701-04-00-02-22.

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TABLE I. - AVERAGE COMPONENT REJECTED ENERGIES

Component	Rejected energy, kW
Mercury pump bearings	0.3
Mercury pump motor heat exchanger	.2
Mercury pump space seal heat exchanger	.5
Primary loop pump heat exchanger	1.2
Heat rejection loop pump heat exchanger	1.2
Turbine space seal heat exchanger	4.8
Alternator heat exchanger	2.2
Turbine alternator bearings	5.0
Total	15.4

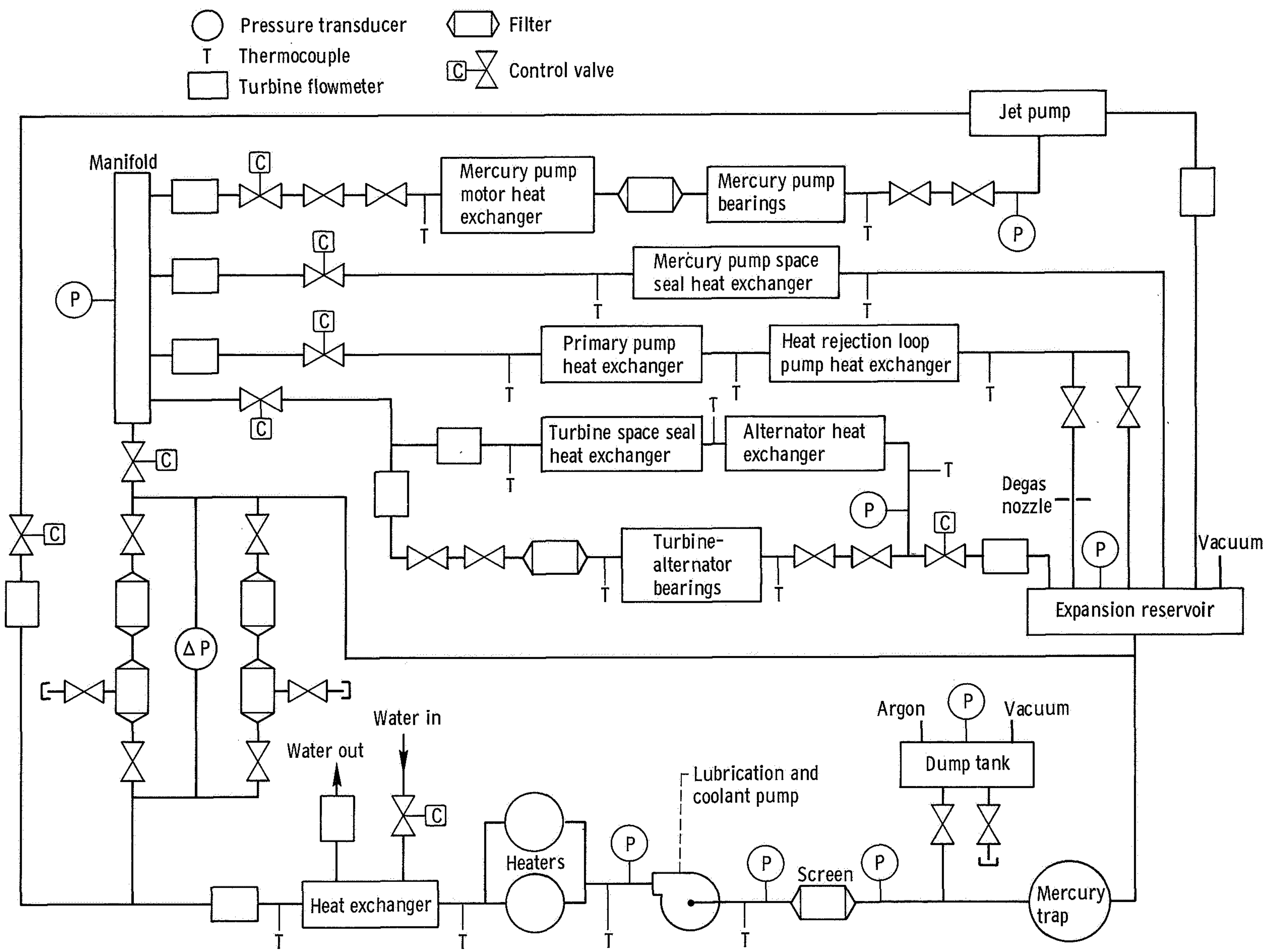


Figure 1. - Lubrication and coolant system schematic drawing.

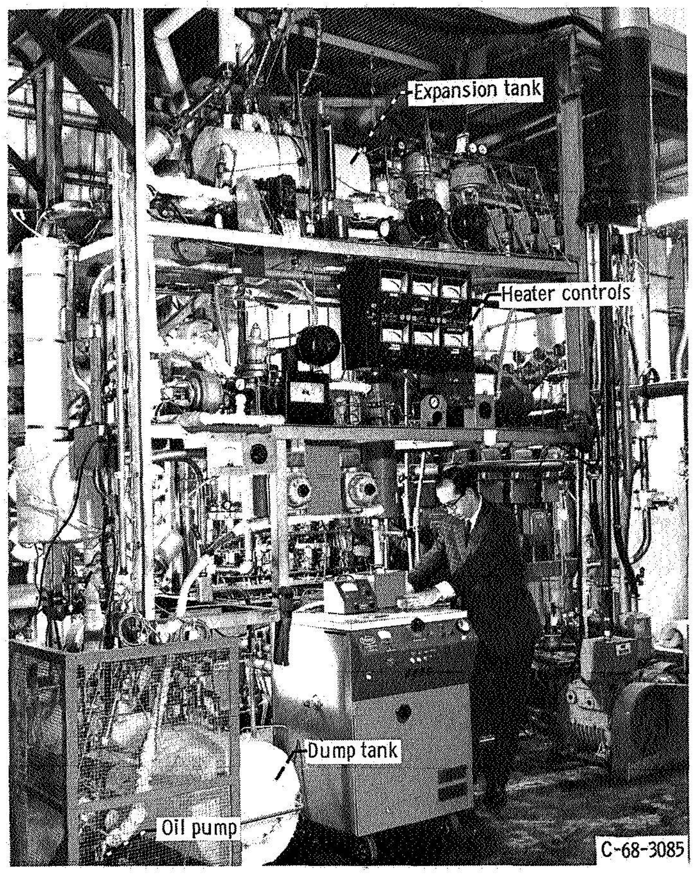
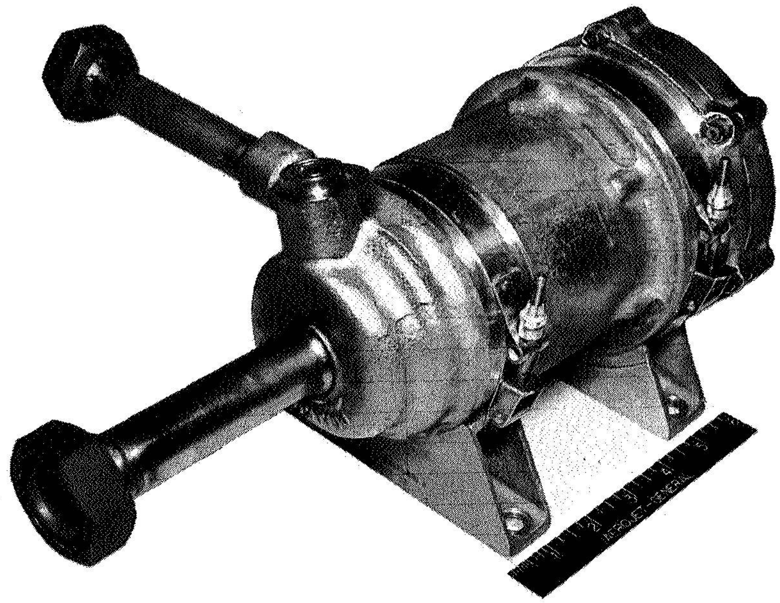


Figure 2. - Overall lubrication and coolant system.



(a) External view.

Figure 3. - Lubrication and coolant pump.

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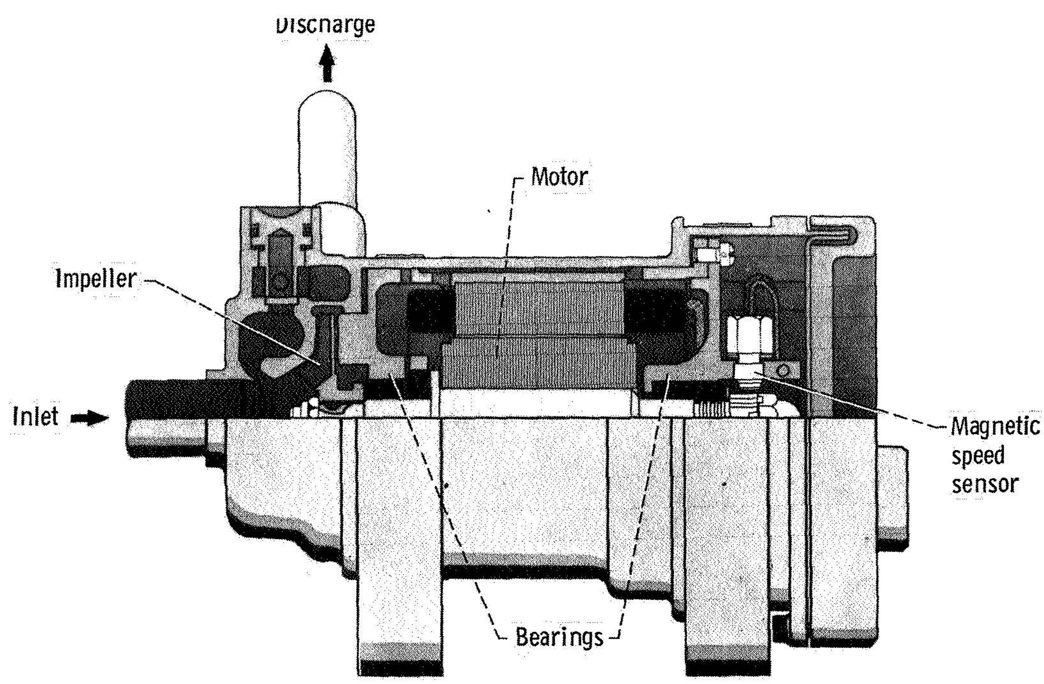

(b) Sectional view.

Figure 3. - Continued.

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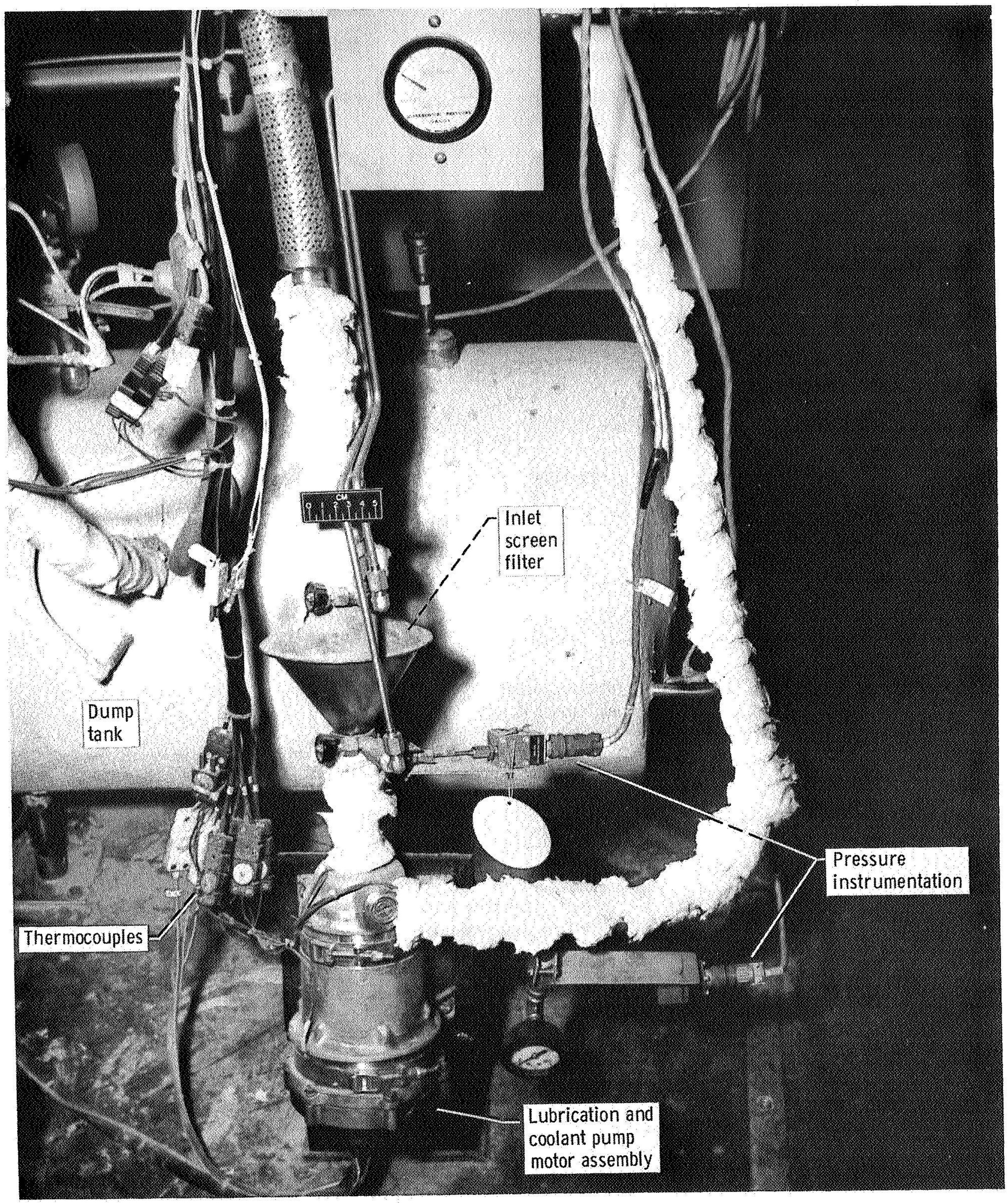

(c) Installation.

Figure 3. - Concluded.

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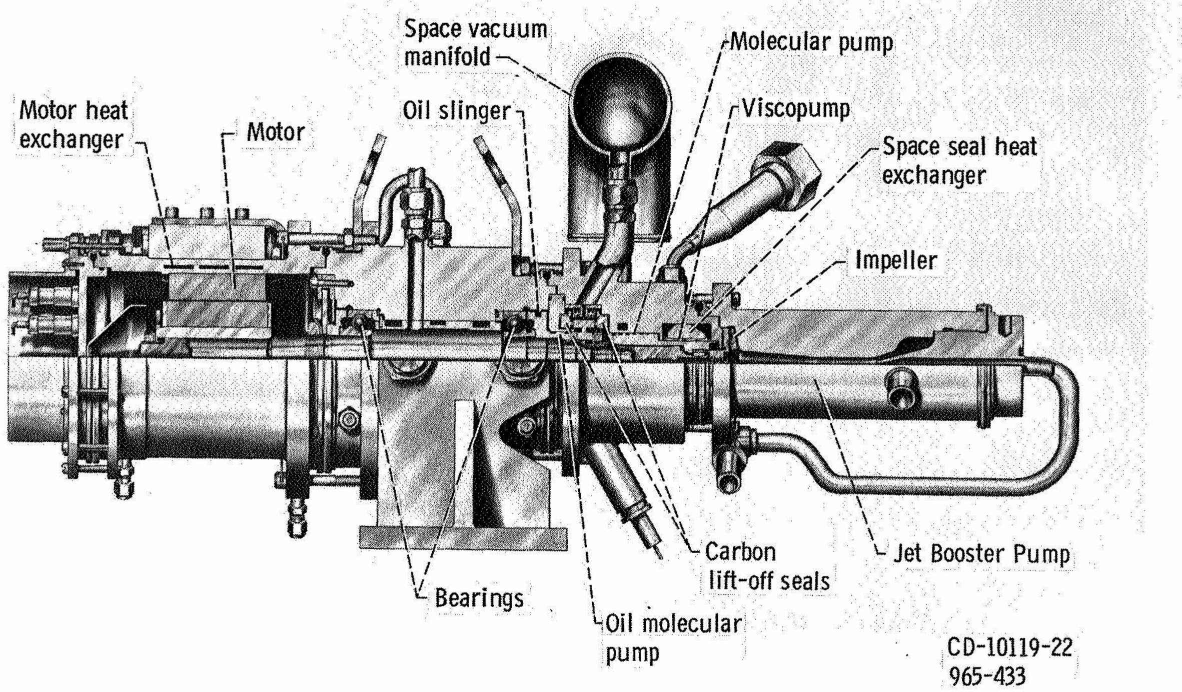


Figure 4. - Mercury pump sectional view.

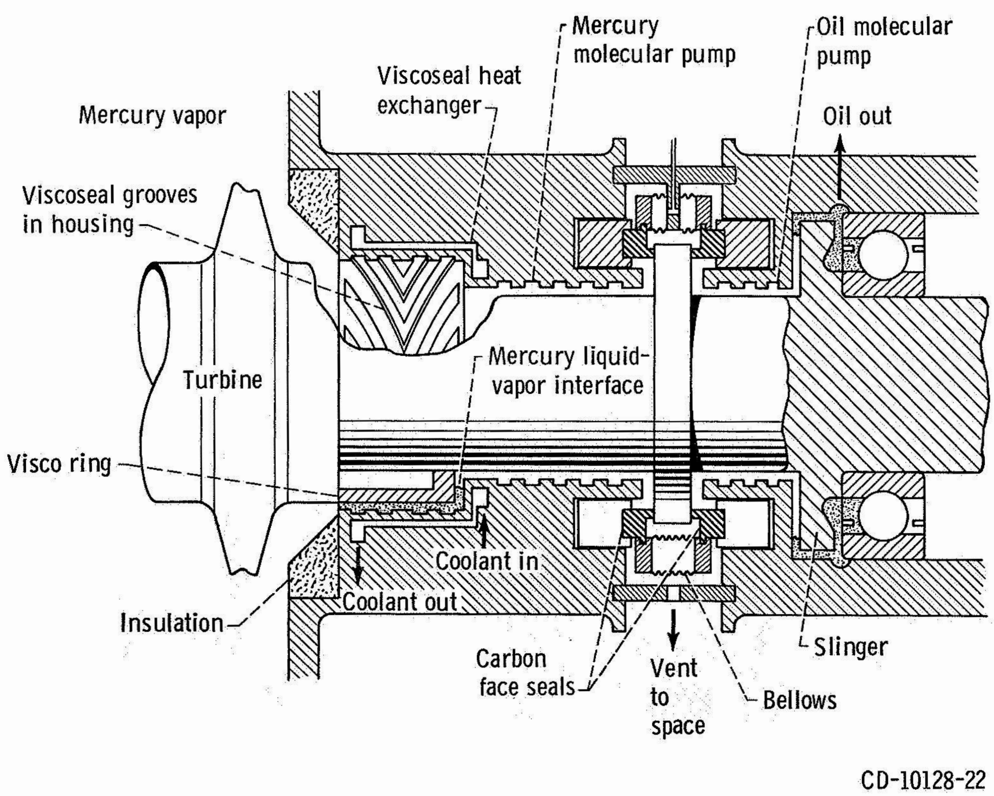


Figure 5. - Turbine-alternator space seal.

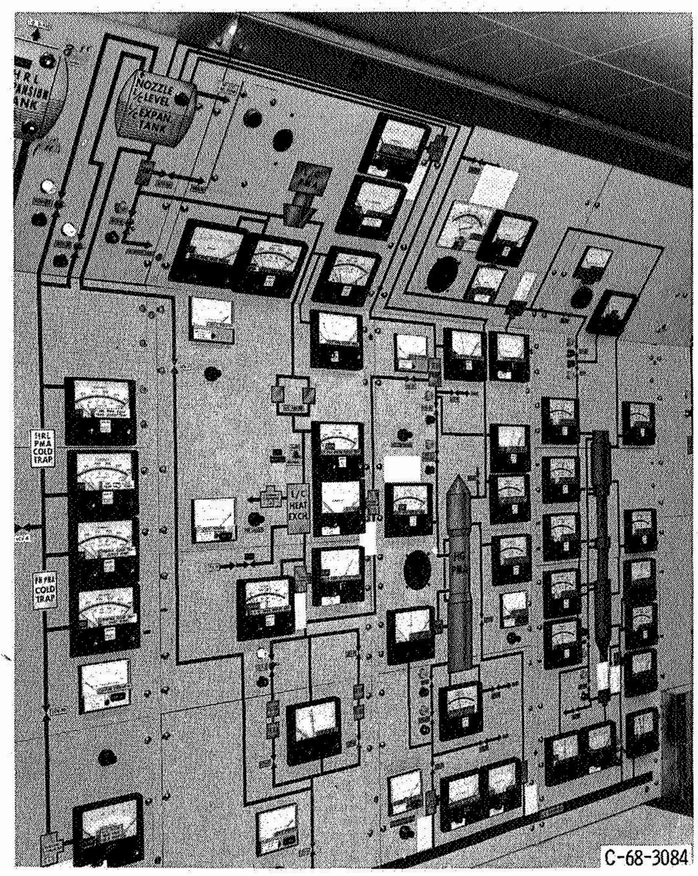
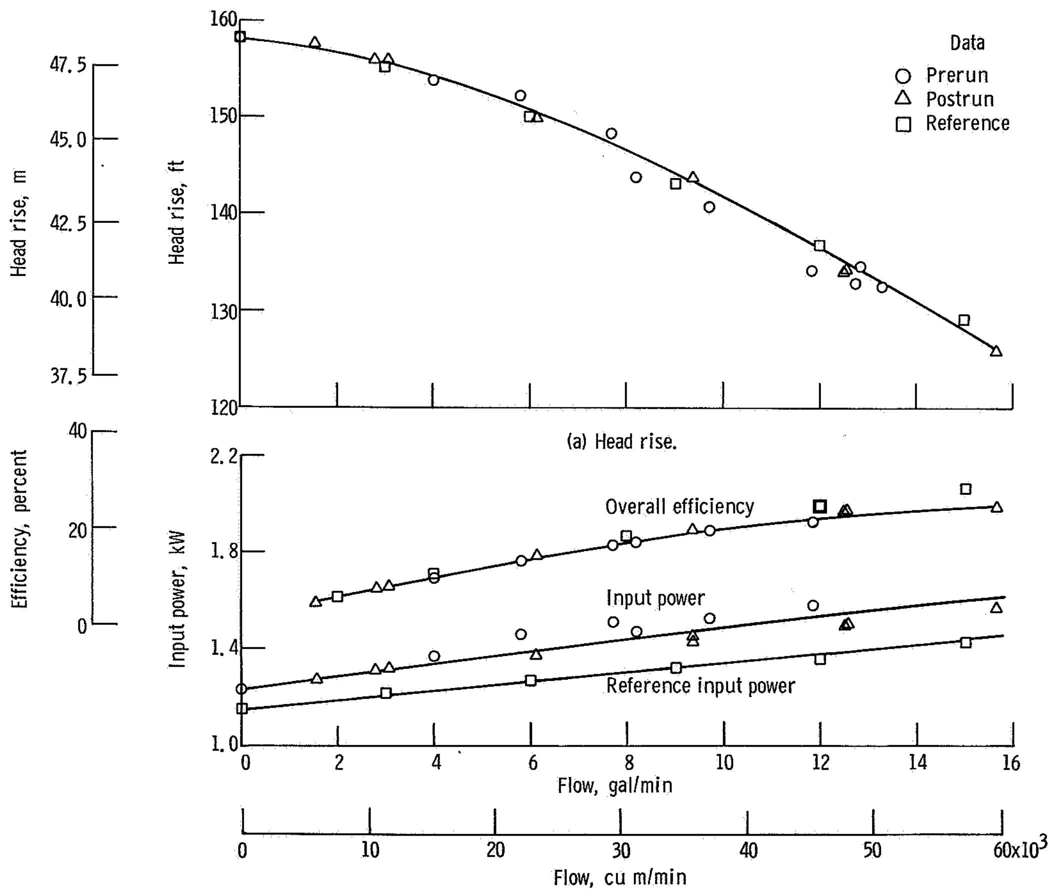
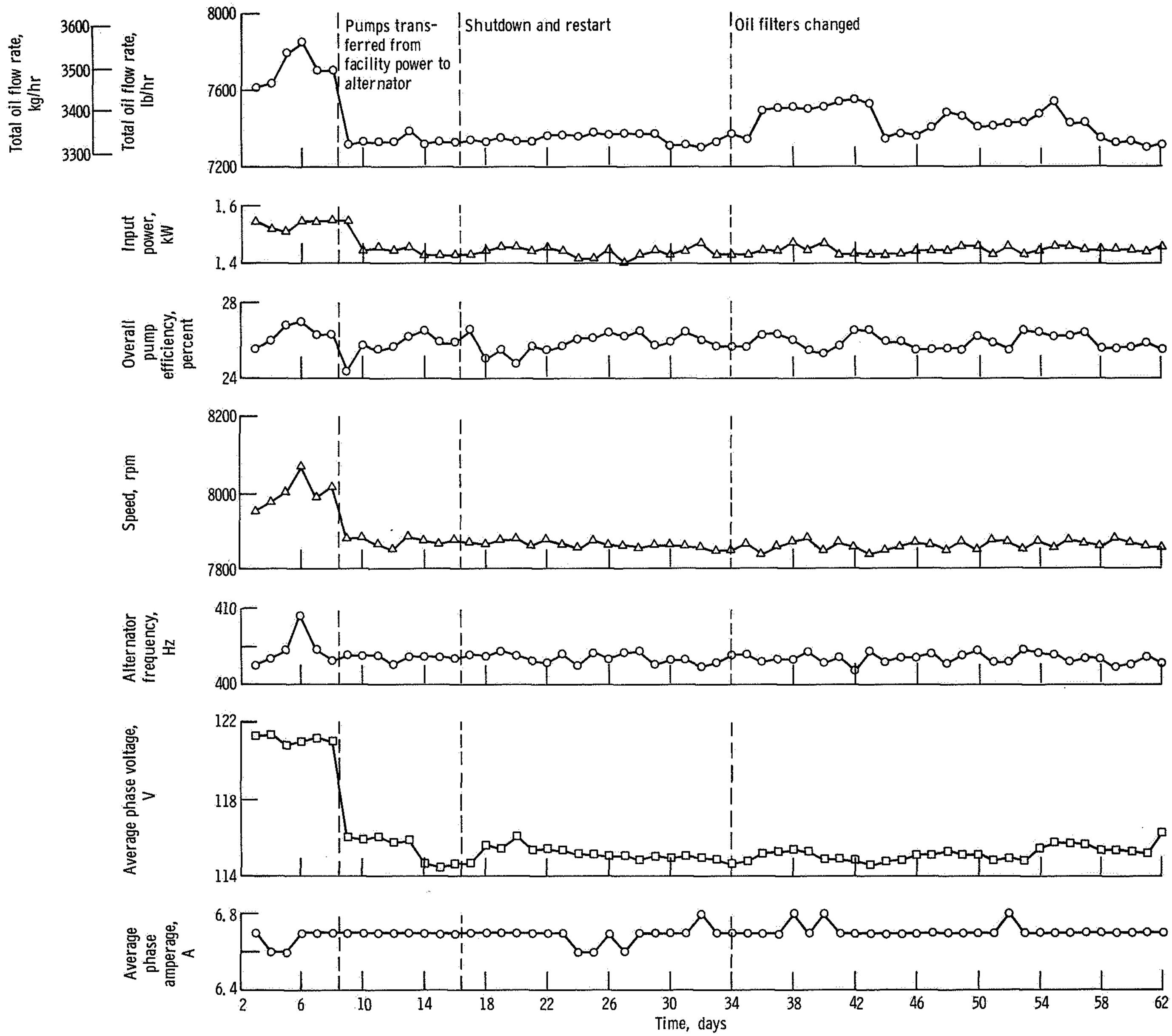


Figure 6. - Control room.



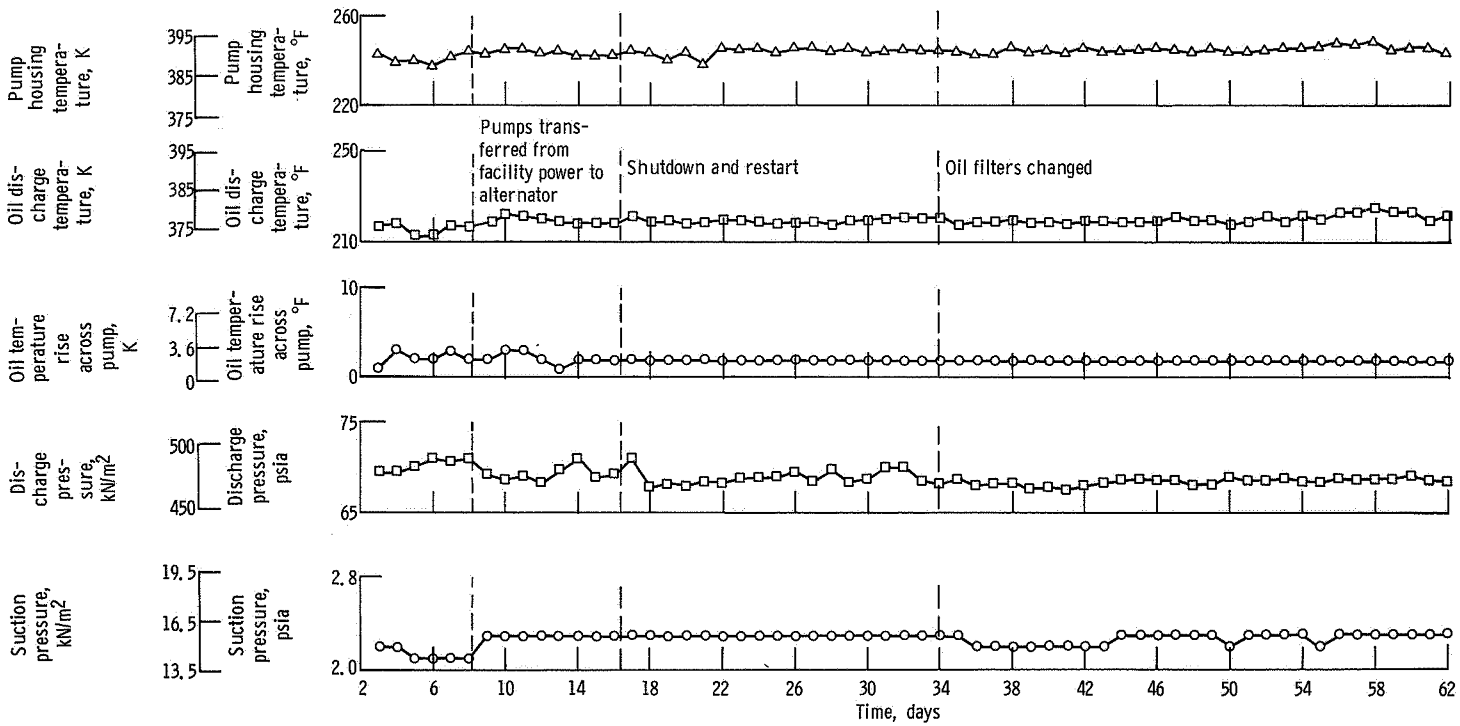
(b) Power and efficiency.

Figure 7. - Lubrication and coolant pump performance curves.



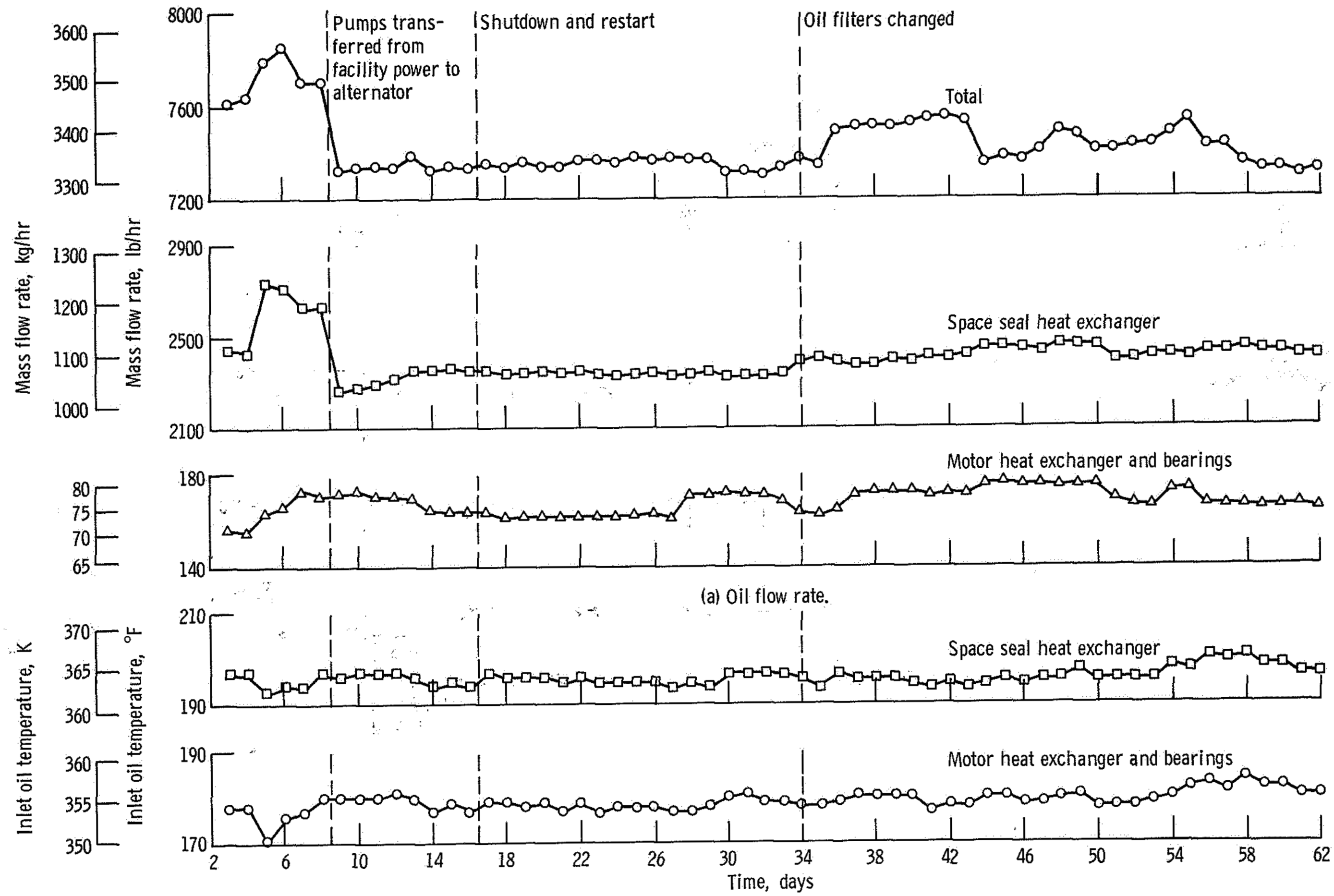
(a) Electrical.

Figure 8. - Lubrication and coolant pump parameters.



(b) Mechanical.

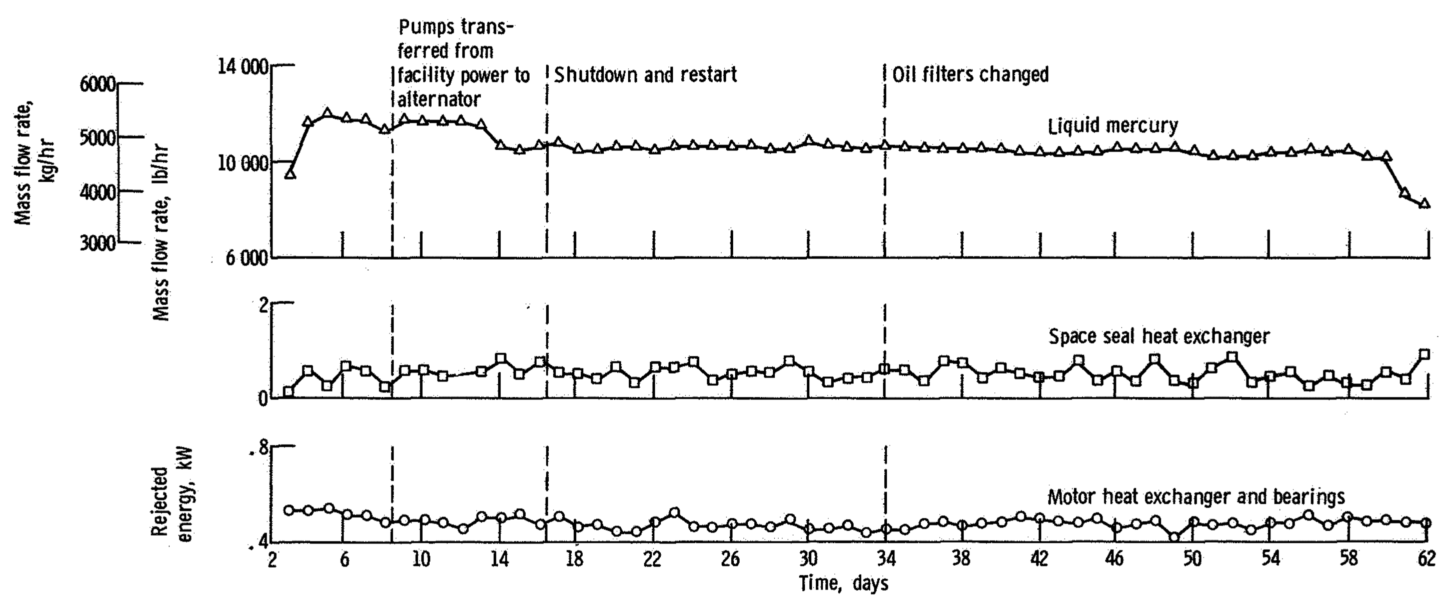
Figure 8. - Concluded.



(a) Oil flow rate.

(b) Supply oil temperature.

Figure 9. - Mercury pump endurance data.



(c) Rejected energy.

Figure 9. - Concluded.

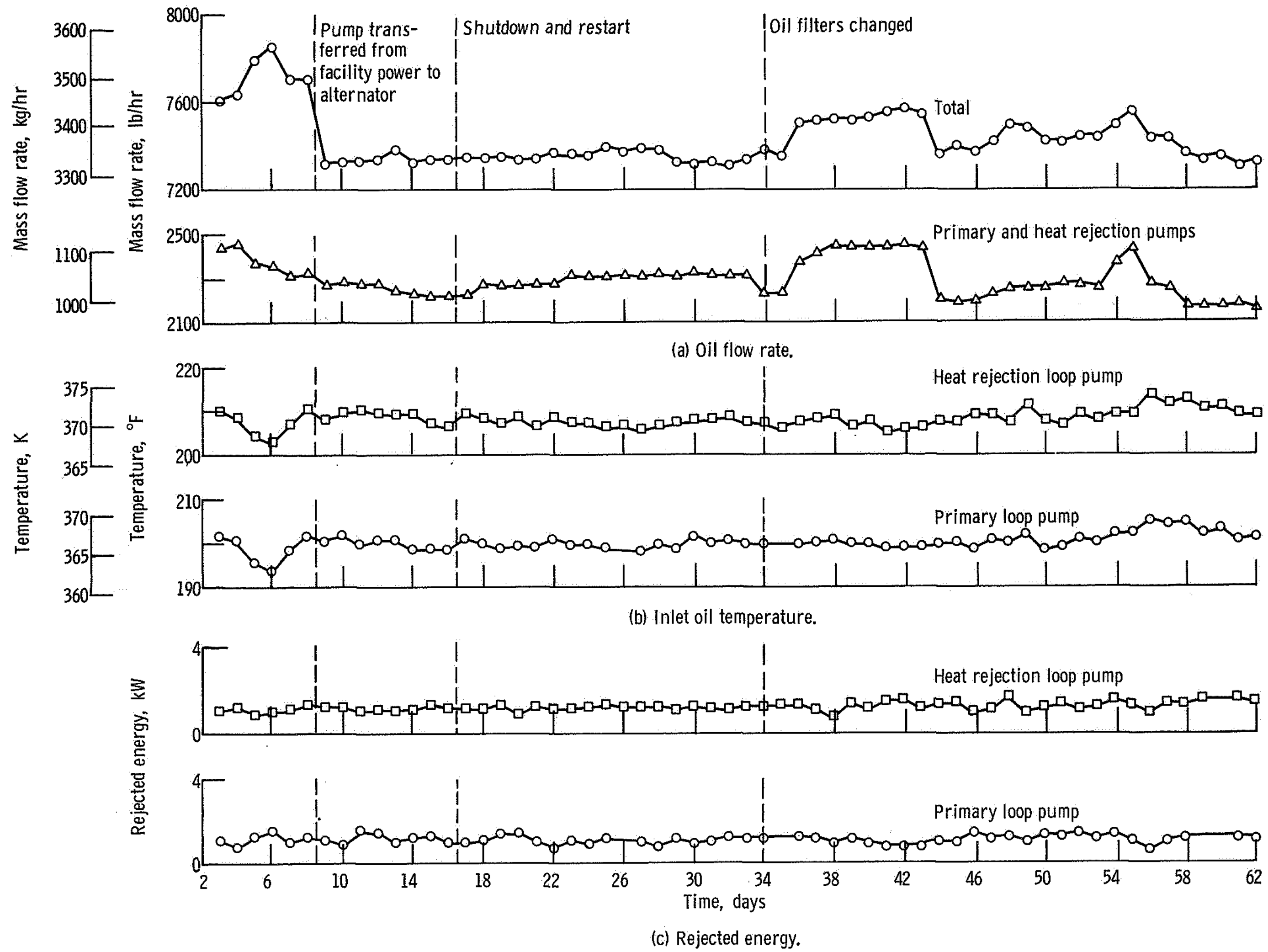
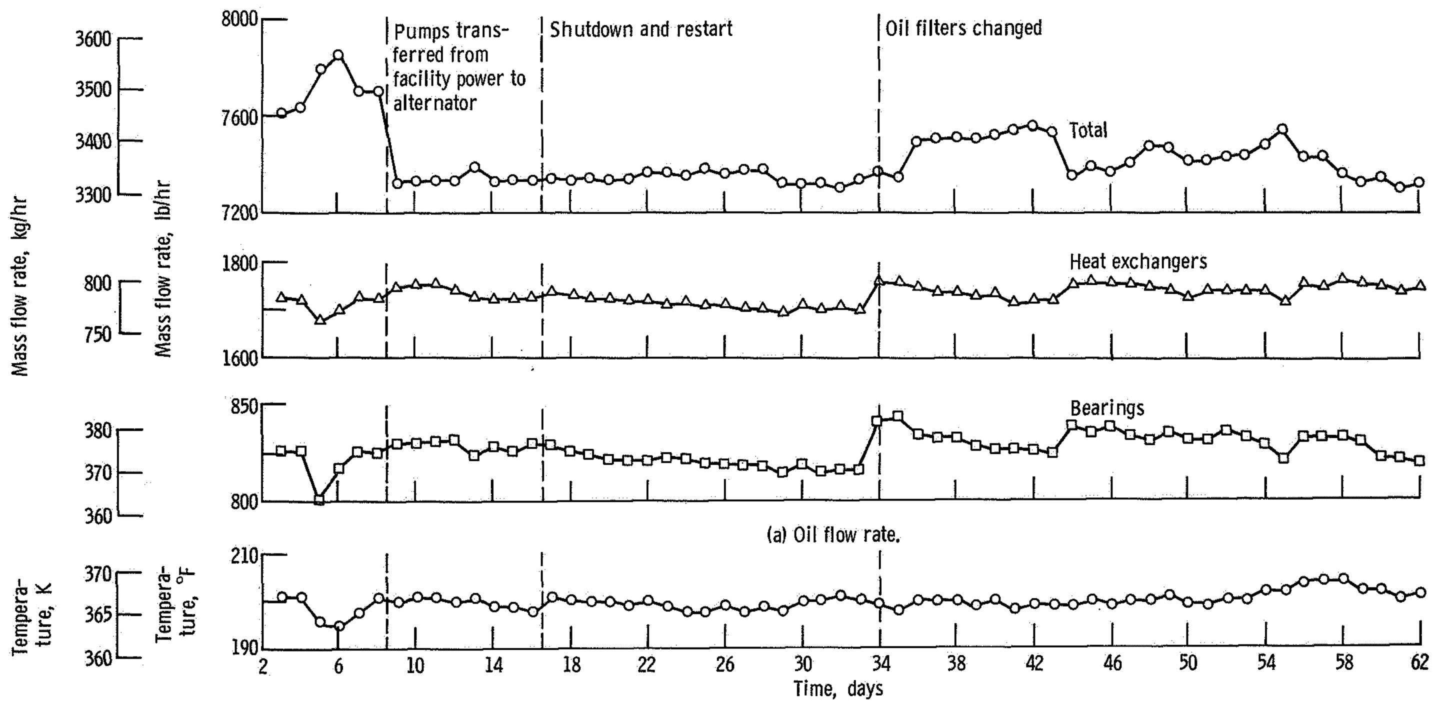


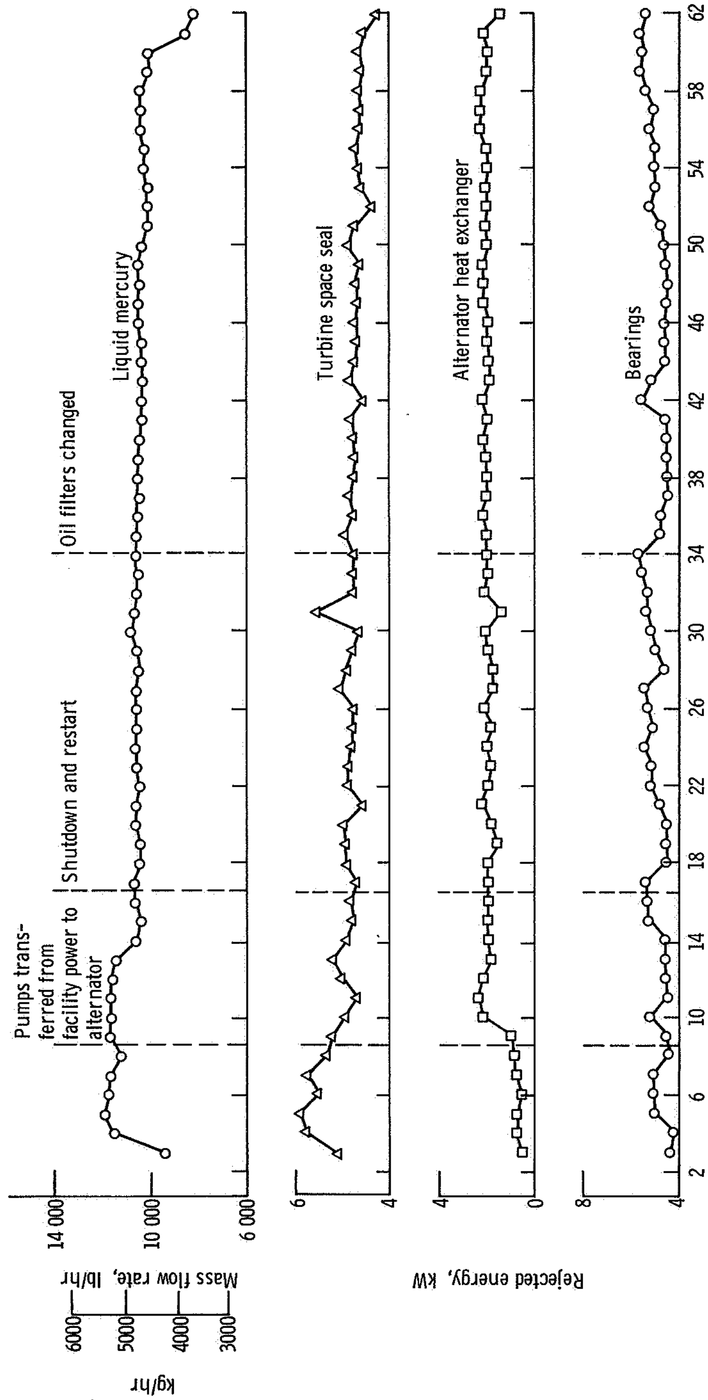
Figure 10. - NaK pumps endurance data.



(a) Oil flow rate.

(b) Supply oil temperature.

Figure 11. - Turbine-alternator endurance data.



(c) Rejected energy.

Figure 11. - Concluded.

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—NATIONAL AERONAUTICS AND SPACE ACT OF 1958

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